Applied Thermal Engineering 71 (2014) 481–500

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Numerical analysis of charging and discharging performance of a thermal energy storage system with encapsulated phase change material

Selvan Bellan [a], José Gonzalez-Aguilar [a, *], Manuel Romero [a], Muhammad M. Rahman [b, c], D. Yogi Goswami [b, d], Elias K. Stefanakos [b, e], David Couling [f]

[a] IMDEA Energy Institute, Ramón de la Sagra 3, 28935 Móstoles, Spain
[b] Clean Energy Research Center, University of South Florida, Tampa, FL, USA
[c] Department of Mechanical Engineering, University of South Florida, Tampa, FL, USA
[d] Department of Chemical & Biomedical Engineering, University of South Florida, Tampa, FL, USA
[e] Department of Electrical Engineering, University of South Florida, Tampa, FL, USA
[f] E.ON New Build & Technology Limited, Nottingham NG11 0EE, UK

HIGHLIGHTS

- Numerical analysis on charging and discharging performance of a TES system.
- Influence of operating parameters and system configuration on melting and solidification processes.
- Numerical modeling of the TES system to elucidate its performance.
- Dynamic behavior of the system subjected to partial charging and discharging cycles.

ARTICLE INFO

Article history:
Received 4 February 2014
Accepted 5 July 2014
Available online 12 July 2014

Keywords:
Latent thermal energy storage
Thermocline system
Encapsulated phase change material
Molten salt
Concentrating solar power

ABSTRACT

The objective of this paper is to develop a two dimensional two-phase model to study the dynamic behavior of a packed bed thermal energy storage system, which is composed of spherical capsules of encapsulated phase change material (PCM-sodium nitrate) and high temperature synthetic oil (Therminol 66) as heat transfer fluid. The heat transfer coefficient is calculated based on the phase change process inside the capsule by enthalpy formulation model and the flow inside the system is predicted by solving the extended Brinkman equation. After model validation, the developed model is used to investigate the influence of capsule size, fluid temperature (Stefan number), tank size (length and diameter), fluid flow rate and the insulation layer thickness of tank wall on the performance of the system. The dynamic behavior of the system, subjected to partial charging and discharging cycles, is also analyzed. It is found that increasing the capsule size, fluid flow rate, or decreasing the Stefan number, results in an increase in the thermocline region which finally decreases the effective discharge time and the total utilization.

1. Introduction

Concentrating solar power (CSP) technologies have been projected as one of the most promising candidates for substituting conventional power generation technologies [1]. Although it is variable as most of the renewable energy systems, like solar photovoltaic and wind, due to the sunlight availability, clouds, aerosol, etc., it can be coupled with a thermal energy storage system (TES), which stores the solar thermal energy for later use and increases the energy source availability beyond normal daylight hours. Hence, it can significantly increase the hours of electricity generation and improve the dispatchability of CSP plants. Basically, thermal energy can be stored in three methods: sensible, latent and thermo-chemical heat storages. Various TES systems have been proposed and implemented in the past few decades [2].

---

[*] Corresponding author.
E-mail addresses: selvanplasma@gmail.com, selvan.bellan@imdea.org (S. Bellan), jose.gonzalez@imdea.org (J. Gonzalez-Aguilar).

http://dx.doi.org/10.1016/j.applthermaleng.2014.07.009
1359-4311/

Several works on the sensible heat storage in packed beds are found in the literature [3–5]. As a pioneering work, Schumann [3] presented the first numerical study on modeling of the packed bed, which has been widely adopted in subsequent studies. The temporal variation of heat transfer fluid (HTF) and filler bed temperatures at the axial symmetry of the tank is predicted by this model. The performance of thermal storage system filled with quartzite rocks, for parabolic trough CSP plants, was investigated by Yang and Garimella [6] using a CFD model, and then the cyclic behavior of sensible thermocline storage system was investigated and found that the cycle efficiency was intimately influenced by the filler particle diameter and radius of the tank for a given mass flow rate [4]. Researchers in the National Renewable Energy Laboratory (NREL) numerically modeled the packed-bed molten salt thermocline system in which the solid fillers were in the form of hexagonal rods or a honeycomb-like structure [7]. Using the adopted Schumann model [3], the performance of thermocline energy storage system, filled with rocks as filler material, was studied by Van Lew et al. [8] and the effects of particle diameter, bed dimensions, fluid flow rate and the solid filler material on the dynamic performance of thermocline storage system were studied by Hänchen et al. [9].

Storing the thermal energy in the form of latent heat of fusion of a phase change material (PCM) significantly increases the energy density, thus potentially reduces the storage size and cost compared to the sensible heat storage system. Hence, several authors have been focused on the use of phase change materials for thermal energy storage. Experimental and numerical studies have been conducted to characterize the PCMs [10,11]. Various studies on the latent heat storage in packed beds have been found in the literature [12–14]. Theoretical and experimental investigations on the transient thermal characteristics of a phase-change thermal energy storage system packed with spherical capsules were conducted by Saitoh and Hirose [12], the influence of capsule diameter and the fluid flow rate on the overall thermal response of the TES was studied. Brief reviews of the work performed on thermocline storage system with PCM capsules were presented [13,14]. The dynamic discharging characteristics of TES system with coil pipes were studied by Ref. [15]. The n-tetradecane was taken as PCM and the aqueous ethylene glycol solution with 25% volumetric concentration was used as heat transfer fluid and the influence of the inlet temperature of HTF, flow rate and the diameter of coil pipes on the outlet temperature of the heat transfer fluid was analyzed. An experimental study was carried out to evaluate the thermal behavior of different TES units coupling with a micro-CHP system [16]; a cylindrical TES tank was used to compare the performance of two phase change materials with different melting temperature and encapsulation method. The mathematical models reported in the literature can be subdivided into three major groups: Single phase model [17], continuous solid phase model [18] and the concentric dispersion model [19]. Comparing the utilization of these three models the continuous solid phase model is more convenient than the concentric dispersion model and more accurate than the single phase model, thus it has been extensively used to study the thermal performance of the packed bed systems [20].

Although several studies are reported on a thermocline system packed with sensible filler materials, the literature on the performance of a latent thermocline energy storage system is relatively few; paraffin is often used as a phase change material, given its ease of handling and adoption to low temperatures building applications. Moreover, the heat transfer mechanisms that govern the charging and discharging processes at high temperature are still under development. Previous studies lack in the high temperature phase change process applications and a clear knowledge about the heat transfer coefficient between the HTF and PCM capsules during charging and discharging processes. Accordingly, this study aims at developing a two dimensional two phase model (continuous solid phase) to analyze the dynamic behavior of a packed bed latent thermal energy storage (LTES) system for high temperature applications. The LTES system is filled with spherical capsules. Sodium nitrate and Therminol 66 are used as PCM and HTF respectively. The developed model is used to predict the heat transfer coefficient between the HTF and the PCM capsules (based on the system configuration) and to study the influence of capsule size, fluid temperature (Stefan number), tank size (length and diameter), HTF flow rate and insulation layer thickness on the performance of the system.

2. Model description

2.1. Governing equations

Fig. 1(a) shows the schematic of a lab-scale thermal storage tank; height L and radius R, packed with PCM encapsulated spherical capsules. The average porosity of the tank is defined as $\varepsilon_{\text{avg}} = V_{\text{HTF}}/V_{\text{tank}}$. During the charge (discharge) process, the hot (cold) thermal oil enters the storage tank at inlet, exchanges the heat with the PCM encapsulated capsules, and leaves the tank via the outlet with a lower (higher) temperature. The two phase model is developed based on the following assumptions:

(1) The thermo-fluid flow is assumed as symmetrical about the axis, thus, the governing equations for heat transfer and fluid flow within the storage tank become two-dimensional.
(2) The PCM capsules behave as continuous, homogeneous and isotropic porous medium.
(3) The flow inside the tank is incompressible, and the radiation heat transfer between the capsules is negligible.
(4) The rhombic packing is assumed. Hence, the distribution of spherical capsules inside the tank is defined by the porosity (ε) and it varies along the radial direction. A monochromatic exponential expression is used to define the porosity along the radial direction as given below [18]

$$\varepsilon(r) = \varepsilon_{\text{avg}}\left[1 + \left(\frac{0.87}{\varepsilon_{\text{avg}}} - 1\right)\exp\left(-5\frac{R-r}{d}\right)\right] \tag{1}$$

where d is diameter of the capsule. Since the Reynolds number is low (less than 380), the flow is assumed as laminar. Most of the low Reynolds number flows inside the packed bed of spherical capsules are modeled as laminar and validated with experimental results e.g. Ref. [18]. Furthermore, Xia et al. [20] compared the two models, with and without turbulence; there is no significant difference between the results even for the Reynolds number of 8300. Based on the foregoing assumptions, the governing equations for the heat transfer and fluid flow inside the tank are given as follows; the axial velocity along the radial direction is obtained by solving the extended Brinkman equation;

$$\frac{\partial P}{\partial z} = -A\frac{(1-\varepsilon^2)}{\varepsilon^3}\frac{\eta}{d^2}u(r) - B\frac{1-\varepsilon}{\varepsilon^3}\frac{\rho}{d}u(r)^2 + \frac{\eta_{\text{eff}}}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right) \tag{2}$$

where η, ρ, P, u and η_{eff} are dynamic viscosity, density, pressure, velocity and the effective viscosity respectively. Due to the wall friction, the velocity gradient near the wall is a function of effective viscosity which is calculated by the following equation [21]

$$\frac{\eta_{\text{eff}}}{\eta} = a\exp(b\text{Re}) \tag{3}$$